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# Comparative study on Gasoline-Ethanol Dual Fuel Injection Strategies in a Small Spark Ignition Engine

Nizar F.O. Al-Muhsen*, Guang Hong
School of Electrical, Mechanical and Mechatronic Systems
University of Technology Sydney (UTS), NSW, 2007, Australia

**Abstract**

Combined ethanol as oxygenated fuel with gasoline fuel can have significant synergies with a small spark ignition engines (SSIE) due to the higher ethanol's latent heat of vaporization, laminar flame speed and research octane number than that of gasoline. This paper reports an experimental investigation to the effect of three dual injection strategies, gasoline port injection (GPI) plus ethanol direct injection (EDI), GPI plus gasoline direct injection (GDI) and GPI only, on the engine performance and the emissions. Experiments were conducted on a modified 249cm3 single cylinder spark ignition engine at a stoichiometric air-fuel ratio ($\lambda$=1), constant engine speed and medium engine load. Five injection volumetric percentages were chosen; 0, 25, 50, 75 and 100%. Among the three injection strategies EDI+GPI demonstrated the best engine performance in terms of the increase of IMEP and nitric oxide emission reduction and GPI the second. However, when the percentage of the fuel directly injected increased to 44% for ethanol and 76% to gasoline, the combustion becomes more unstable and more CO and HC emitted than that in GPI only.

*Keyword: Dual injection, Ethanol direct injection, Gasoline direct injection, Port injection.*

## 1. Introduction

Developing a recent fossil fuel or adopting an alternative environmentally friendly one is a common topic among the worldwide researchers in the last few years. Recently, ethyl alcohol (Ethanol) as an oxygenated biofuel has been seen as an attractive fuel that can improve the gasoline performance or even replace it [1-4]. Practically, ethanol fuel can help to improve the combustion quality and its efficiency that will lead to enhance the engine thermal efficiency, engine capacity and emissions [5-7]. This can be mainly attributed to the ethanol's excellent properties such as the latent heat of vaporization, octane number and laminar burning velocity if it is compared with gasoline fuel as shown in table1.

The dual injection strategy is a new technology that utilized to optimize the spark ignition engine performance providing more flexibility to the direct fuel injection to port fuel injection ratio that can be immediately changed according to the needed operating conditions. Gasoline direct injection engines has been commercially produced due to its high performance comparing with the conventional port fuel injection engines. However, these engines produce a larger amount of emissions (particular matter) because the mixture before combustion might be non-homogeneous [8, 9]. In contrast, the directly injected ethanol can relatively enrich more completed and less emission combustion due to its remarkable properties [5, 7].

The presented work aims to investigate fuel injection strategies in combination of multiple fuels used to optimize the engine performance, maximize the thermal efficiency and minimize the emissions at a medium engine load of a small SI engine. In order to do that, two injection strategies were investigated and compared with the conventional port fuel injection strategy as a baseline. Moreover, combustion and emissions taking into account as a main two parameters that can indicate to engine performance.

| **Property** | **Unit** | **Gasoline** | **Ethanol** |
|---|---|---|---|
| Chemical formula | - | $C_2$-$C_{12}$ | $C_2H_5OH$ |
| Molecular weight | kg/kmol | 114.15 | 45.07 |
| H/C | Atom ratio | 1.795 | 3 |
| O/C | Atom ratio | 0.7-0.78 | 0.794 |
| Density (at 288.15K) | kg/m$^3$ | 750-765 | 785-809.9 |
| Stoichiometric air-fuel ratio | w/w | 14.2:1-15.1:1 | 8.97:1 |
| Kinematic viscosity | mm$^2$/s | 0.5-0.6 | 1.2-1.5 |
| Octane number | - | 91 | 108.61 |
| Higher heating value (HHV) | MJ/kg | 44.0 | 26.9 |
| Laminar flame speed at 100kPa, 325K | cm/s | ~33 | ~39 |
| Latent heat of vaporization | kJ/kg | 298 | 948 |
| Saturation vapour pressure | kPa | 28.828 | 8.773 |

Table1. Ethanol and Gasoline Fuel Properties [2, 4, 10-12]

## 2. Experimental Setup and Methodology

### 2.1 Test Rig Engine

A 4-stroke single cylinder has been adopted to conduct this experimental work. This spark ignition engine was modified from conventional port fuel injection engine to be equipped with flexible dual fuel injection system that can be controlled by a computer. An engine control system was developed by Hents Technology to adjust and monitor the spark timing, the mass of fuel injected,

* Corresponding author: Nizar F.O. Al-Muhsen
Phone: (+61) 423095545
Email: Nizar.Al-Muhsen@student.uts.edu.au

fuel injection timing and pressure, throttle position and engine speed (RPM). An eddy current dynamometer with control system was coupled to the engine to uphold the engine speed and measure the engine torque. The in-cylinder pressure was recorded using a Kistler 6115B measuring spark plug pressure transducer. More details about the engine test rig can be found in [3].

## 2.2 Experimental Methodology

Three fuel injection strategies: Gasoline port injection only (GPI) which was set as a baseline, EDI+GPI and GDI+GPI were set at about stoichiometric air/fuel ratio ($\lambda$=1), a fixed engine speed of 4000 RPM, spark timing of 15 crank angle degree before top dead center (CAD BTDC) and injection timing of 300CAD BTDC. In different injection strategies and direct injection volumetric percentages (DIVP), the total fuel heating energy (~580J) is kept the same. During the engine tests, methodical comparisons were implemented to the three combustion modes as shown in the matrix test (Table 2) with a focus on the effect of the injection strategy on the engine performance and emissions. The DIVP of fuel was changed from 0% as GPI only, which represented as a baseline for the experimental tests, to 100% as DFI only. This includes ethanol direct injection (EDI) and gasoline direct injection (GDI). The engine was started and warmed up to 200±10$^o$C, as the designated engine operating temperature, with GPI only, and then DIVP gradually increased from 0% to 100%. Five samples were taken for each data and then the average was used in calculations and analyses. The in-cylinder pressure was recorded three times for each 100 consecutive cycles at a rate of 0.5 crank angle degree (CAD) intervals and then averaged to be counted into the theoretical calculations.

| Test | Symbol | Port injection | Direct injection | DIVP% |
|---|---|---|---|---|
| Baseline | GPI only | Gasoline | - | 0 |
| Strategy1 | GPI+EDI | Gasoline | Ethanol | 0-100 |
| Strategy2 | GPI+GDI | Gasoline | Gasoline | 0-100 |

Table2. Test matrix

# 3. Results and Discussion

Results of engine performance and emissions are represented and discussed to show the effect of different fuel injection strategies.

## 3.1 Engine Performance

Figure 1 shows the variation of IMEP with DIVP at 4000 RPM. It compares the three injection strategies GPI only, EDI+GPI and GDI+GPI. As shown in Figure 1, the IMEP increases significantly once the EDI starts at DIVP of 25% ethanol fuel and reaches the maximum IMEP at DIVP of 75% ethanol fuel. This result could be attributed to two main reasons. Firstly, the Oxygen content of ethanol fuel might improve the combustion phasing resulting in higher IMEP comparing with the other two injection strategies. Secondly, the higher ethanol laminar burning velocity than that of gasoline fuel might play a significant role in improving the combustion quality resulting in IMEP enhancement [3, 13, 14]. In contrast, at DIVP of 100% ethanol fuel, the IMEP is reduced. This could be attributed to lower heating value of ethanol coupled with overcharge cooling effect and lower saturation vapor pressure resulting in reducing the heat released and poor mixture quality around the vicinity of the spark plug at the moment of the spark discharge [7].

Concerning GDI, the reduction in IMEP can be attributed to two main reasons. Firstly, as reported by Zhu et al. [15], the IMEP reduction was due to the over cooling effect of GDI that may negatively impact on the IMEP. Secondly, the high-pressure injector’s position might suite EDI but not GDI. Direct injection of gasoline fuel in this engine might cause a higher fuel impingement into the combustion chamber surfaces resulting in higher oil film formation and thus low mixture quality which might cause the IMEP reduction.

Figure 2 shows the variation of DIVP with the coefficient of variation of IMEP ($COV_{IMEP}$) which denotes to the cyclic variability of the engine that must not exceed 10% [16]. As shown in Figure 2, the $COV_{IMEP}$ in the dual fuel injection strategies is smaller than that of GPI only until the DIVP reaches 76% in GPI+EDI and 44% in GPI+GDI.

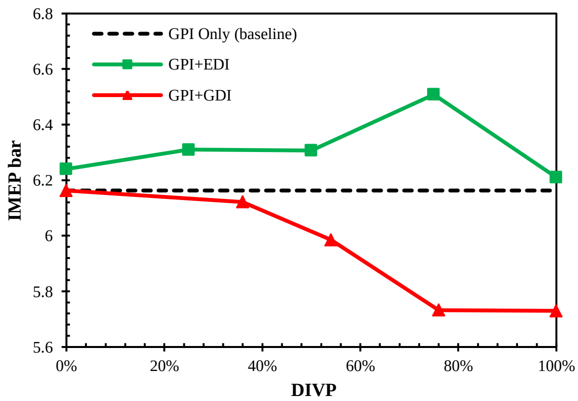


Fig. 1 Variation of IMEP with DIVP

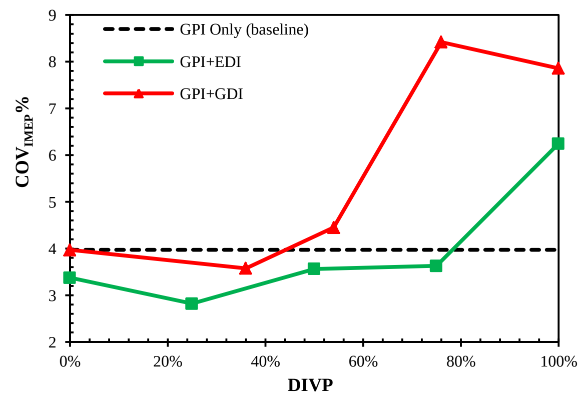


Fig. 2 Variation of $COV_{IMEP}$% with DIVP

This can be attributed to the greater ethanol latent heat of vaporization and burning velocity that may reduce the combustion temperature and shorten the combustion duration respectively. However, for GDI injection, the $COV_{IMEP}$ increased dramatically when the amount of DIVP moved from 54% to 100%. It is assumed that, the high amount of the impinged fuel might cause a more oil film formation on the walls of the combustion chamber resulting in less homogenous and leaner mixture which it is also believed that it might increase the engine cycle-to-cycle variations [16]. Furthermore, $COV_{IMEP}$ increased when EDI move from 75% to 100%. This mainly because the 100% EDI could cause an overcharge cooling effect and thus further increasing of combustion duration resulting in more unstable combustion [15].

Figure 3 shows that the engine volumetric efficiency is slightly improved with the increase of the EDI percentage gradually. It is assumed that inject ethanol directly into the combustion chamber may reduce the mixture temperature and increase its density [3, 14]. However, turning up into 100% EDI can adversely affect the engine volumetric efficiency compared with 75% EDI, possibly due to the combustion deterioration that may happen as a results of the over cooling effect. In contrast, the volumetric efficiency in GDI+GPI is slightly lower than that in GPI only. This may be due to the inferior mixture quality compared with the baseline injection strategy (GPI only).

Figure 4 shows the cylinder pressure in four conditions: GPI only, EDI only, GDI only and 50%EDI+50%GPI. As shown in Fig. 4, the charge cooling effect and laminar flame speed of ethanol combined with oxygen content may speed up the combustion process and then increase the cylinder pressure [13, 14]. This is consistent with the IMEP results shown in Fig. 1. However, the overcharge cooling effect may negatively affect these processes. This can be observed previously in Fig. 1 when the IMEP increased with specified amount of ethanol directly injected (25%≤ EDI ≤75%) and then starts to be decreased when the overcharge cooling happened at 100% EDI.

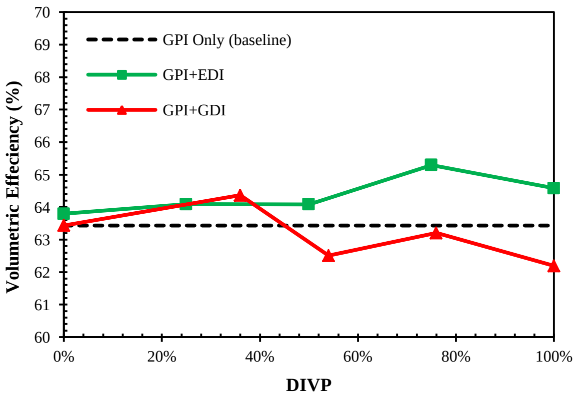


Fig. 3 Variation of the Engine Volumetric Efficiency with DIVP

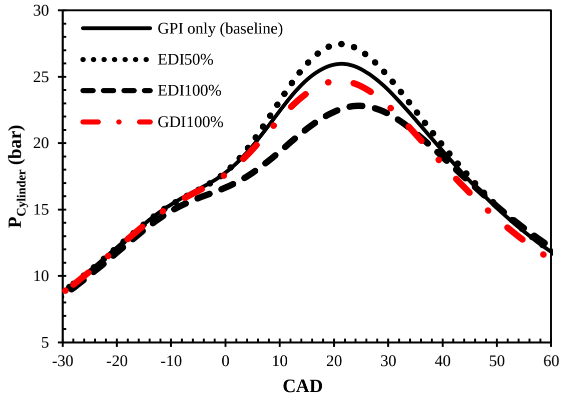


Fig. 4 Variation of Cylinder Pressure with CAD

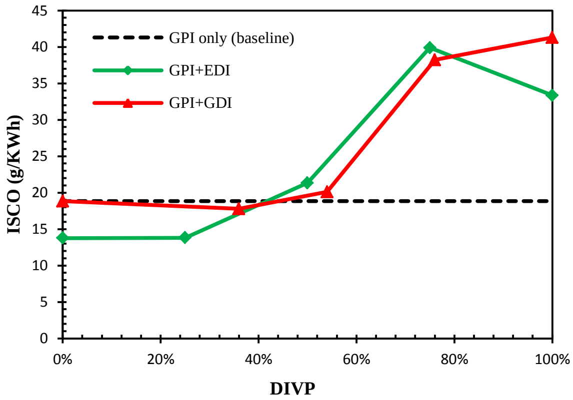


Fig.5 Variation of ISCO with DIVP

## 3.2 Emissions

As shown in Fig. 5 the indicated specific carbon monoxide (ISCO) emissions are slightly reduced with the increase of DIVP. However, once the DIVP is greater than 42%, the ISCO then increases quickly with the increase of DIVP. This result could be attributed to three reasons. Firstly, focusing on ISCO reduction period, ethanol fuel is an oxygenated fuel providing combined with fast laminar flame speed effect could improve the combustion quality resulting in ISCO reduction [7]. Secondly, for both direct injection strategies, when DIVP moves from 42% to 100%, the overcharge cooling effect may reduce the combustion temperature resulting in less ISCO oxidation and thus higher amount of carbon monoxide emissions [16]. Finally, the poor mixture quality after 42% DIVP for both direct injection strategies may adversely affect the combustion quality enlarging the ISCO emissions compared with GPI only.

Figure 6 shows the variation of indicated specific hydrocarbon (ISHC) emissions with DIVP. The ISHC values decrease with increase DIVP reaching a minimum value at 50% in GPI+EDI and at 56% in GPI+GDI and then the ISHC increase reaching the maximum at 100% DIVP. The difference between two strategies is possibly because of the fuel properties. The greater laminar flame speed and the oxygen content of ethanol might contribute to reduce ISHC until 50% of EDI whereas the overcharge cooling effect and lower ethanol volatility may result in lower mixture quality leading to more oil film formation and thus higher amount of ISHC emissions [16]. These can be the main reasons that might contribute increasing ISHC dramatically between 50-100% EDI. In contrast, for GPI plus GDI strategy, the higher gasoline volatility compared with ethanol may play a significant role in the ISHC reduction.

The indicated specific nitric oxide (ISNOx) emissions varied with DIPV are represented in Fig. 7. In fact, the combustion temperature influences directly the NOx formation inside the combustion chamber [17]. It can be clearly seen that the charge cooling effect of both injection strategies in the ISNOx reduction. Actually, ethanol fuel is more efficient in the ISNOx reduction than gasoline. This can be attributed mainly to two reasons. First, as it is mentioned previously, there is more oxygen available to complete the combustion and thus ISNOx reduction. Secondly, the higher latent heat of vaporization might play an essential job in the combustion temperature decreasing resulting in the ISNOx emissions reduction [2].

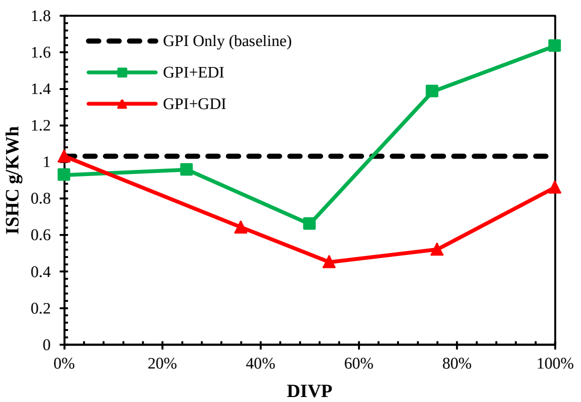


Fig. 6 Variation of ISHC with DIVP

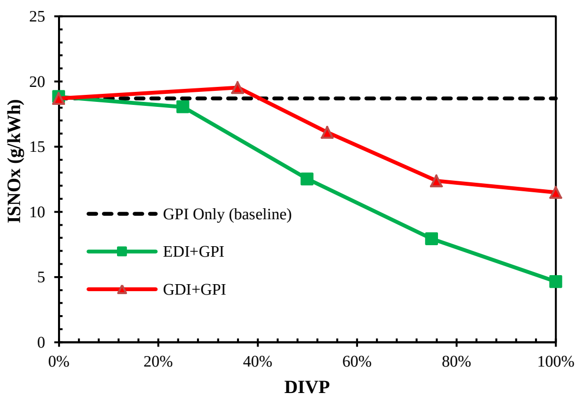


Fig. 7 Variation of ISNOx with DIVP

## 4. Conclusion

An experimental investigation was conducted to investigate dual injection strategies. Experiments were carried out on a small SI engine at GPI only, EDI+GPI and GDI+GPI at medium load and 4000RPM engine speed. The Results of engine performance and emission varied with the percentage of fuel directly injection are presented and discussed. Conclusions can be drawn as follows.

1. Adopting EDI plus GPI strategy improved the combustion quality and thus enlarged IMEP comparing with the other two injection strategies, possibly because of the oxygen contain of ethanol combined with the greater laminar burning velocity compared to that of gasoline.

2. Using GPI plus EDI strategy enhanced the combustion stability by reducing the cycle-to-cycle variation comparing with the other two injection strategies, possibly due to ethanol greater laminar flame speed coupled with the oxygen content.

3. Engine volumetric efficiency slightly improved using GPI plus EDI strategy compared with the other two injection strategies, perhaps due to the high utilizing of ethanol latent heat of vaporization.

4. GPI plus GDI strategy performed better in HC emissions reduction compared with GPI plus EDI strategy, perhaps due to the greater gasoline volatility compared with ethanol fuel.

5. GPI plus EDI strategy behaved better in NOx emissions reduction compared with the other two injection strategies, probably due to the ethanol greater latent heat of vaporization compared with gasoline.

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